Detecting an improperly-lubricated journal bearing in a centrifugal compressor (when your 'toolbox' isn't full)

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Bently Nevada's Machinery Management Services' (MMS) philosophy is to consider all available information when evaluating a machine's health. MMS makes recommendations only after a thorough study of, as a minimum, the *available* vibration, rotor position, process variable, and machinery design information. MMS views a customer's machinery management system as a toolbox full of tools; diagnostic efforts are hampered if tools are missing from the toolbox. This case history discusses one such effort, and presents data acquisition and analysis methods for dealing with toolboxes that aren't full.

High vibration on plant air compressor

The J.R. Simplot Company's nitric acid plant in Helm, California produces 75,000 short tons of nitric acid yearly. The plant air compressor train is vital to the process, providing 27,800 standard cubic feet per minute of air. It is rated at 6100 horsepower and includes a steam turbine driver, a high pressure (HP) and low pressure (LP) centrifugal air compressor, and a hot gas expander (Figure 1). The train operates at speeds near 10,600 rpm.

In the summer of 1993, the air compressor train was overhauled. After the unit was returned to service, steady state direct vibration amplitudes at the high pressure compressor's thrust-end radial bearing (#4) were at 76 μm (3.0 mil) pp. At this point, MMS was contacted to perform onsite diagnostics.

Monitoring system limitations

A Bently Nevada 7200 Monitoring System and *single* prox-

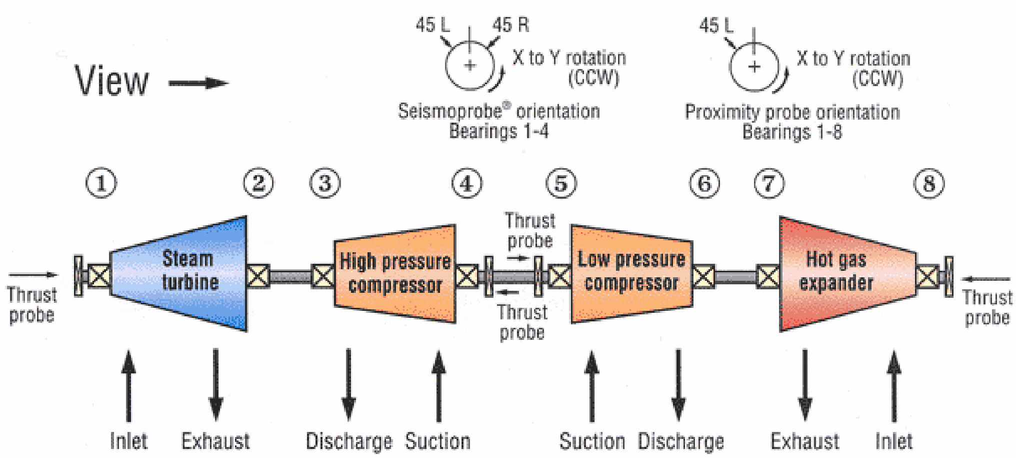

Figure 1
Machinery arrangement diagram.

imity probes mounted at each radial and thrust bearing provided shaft vibration and position information (Figure 1). No Keyphasor® probe was installed to provide a 1X rotor phase reference. The signal from a proximity probe viewing a 60-tooth auxiliary drive gear (providing a 60X pulse) was directed to a tachometer in the control room.

Temporary proximity probes could not be installed due to space limitations and time constraints. The absence of a second, orthogonal proximity probe at each radial bearing meant that shaft orbit, full spectrum, and centerline data were unavailable. Seismic data was acquired with orthogonal Bently Nevada 9200 Seismoprobe® velocity transducers mounted on the turbine and HP compressor's bearing caps. These seismic readings were taken mainly because the alternative was no XY (2 dimensional cross-section) data at all. Seismic data would prove to be of limited use because

1) Shaft orbit, full spectrum, and centerline data can only be generated with XY shaft measurements.

2) Only a fraction of the rotor vibration would be transmitted to the bearing caps due to their high stiffness, damping within the fluid-film bearings, and a high casing-to-rotor mass ratio.

Neither a temporary Keyphasor probe nor optical pickup could be installed; the lack of a once-per-turn Keyphasor signal meant that nX amplitude and phase data were unavailable. A Bently Nevada TK15 Keyphasor Conditioner and Power Supply and TK16 Keyphasor Multiplier/Divider were used to generate a 1X reference pulse from the 60X pulse provided to the control room tachometer. Since the TK16 would trigger on a different reference tooth each run, phase measurements from different runs would not repeat.

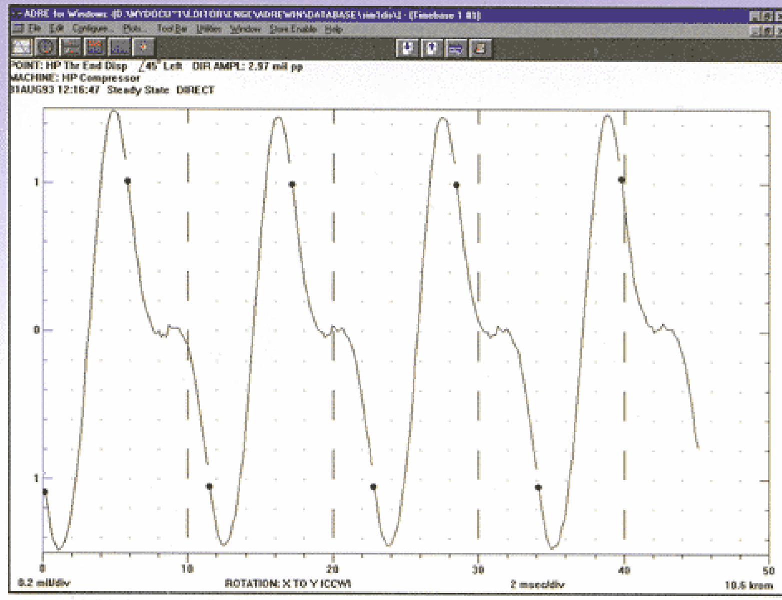

Figure 2
Steady state shaft direct timebase plot, HP compressor thrust end (Bearing #4).

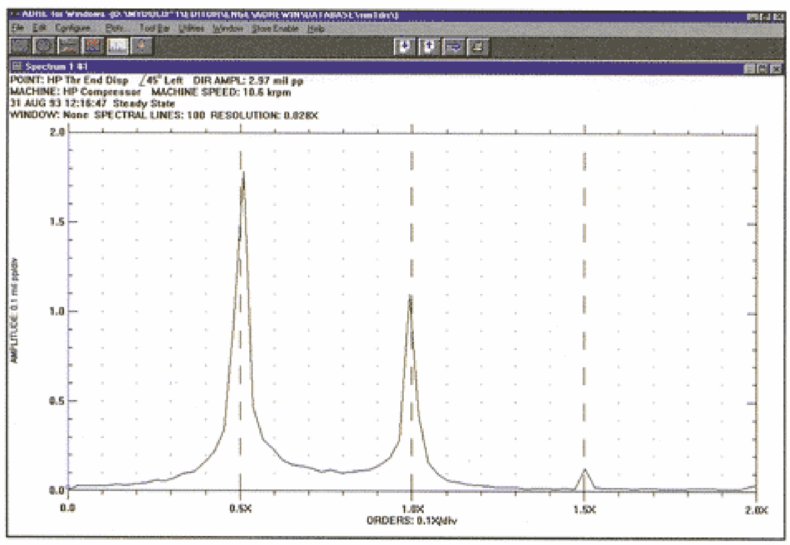

Figure 3
Steady state shaft half spectrum plot, HP compressor thrust end (Bearing #4).

Initial steady state analysis

MMS used a Bently Nevada ADRE®3 System to collect and reduce the vibration data. The ADRE 3 System can simultaneously collect and process vibration data (overall vibration, 1X and 2X amplitude and phase, frequency, rotor position, and waveform). It can acquire data when the machine is at steady speed and load, and also during machine startups and shutdowns. ADRE acquired eight channels of shaft displacement data from the single permanent proximity probes mounted at each bearing, and eight channels of velocity data from the pairs of Seismoprobes mounted on the turbine and high pressure com-

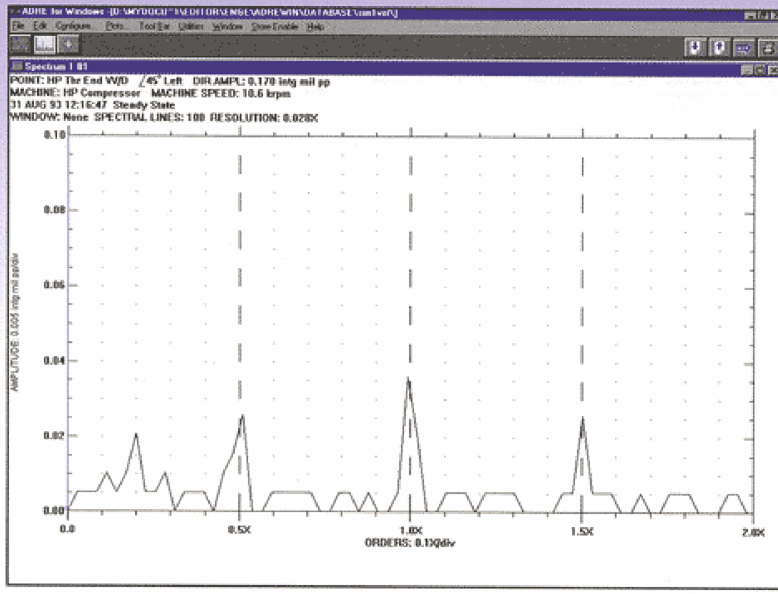

Figure 4
Steady state bearing cap half spectrum plot, HP compressor thrust end (Bearing #4).

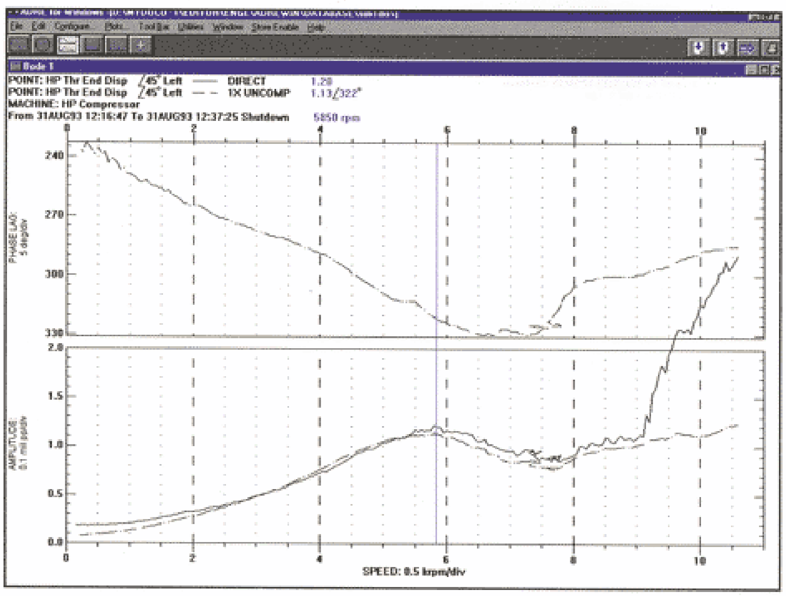

Figure 5
Initial shutdown Bode plot, HP compressor thrust end (Bearing #4).

pressor bearing caps. The velocity readings were electronically integrated to displacement.

Steady state data at 10,595 rpm was acquired before the compressor train was shut down. HP compressor shaft direct amplitudes were 14.2 μm (0.56 mil) pp at the non-thrust end (Bearing #3) and 75.4 μm (2.97 mil) pp at the thrust end (Bearing #4). HP compressor bearing cap direct amplitudes (velocity integrated to displacement) were 3.3 μm (0.13 mil) pp at Bearing #3 and 4.3 μm (0.17 mil) pp at Bearing #4. Figures 2-4 illustrate steady state shaft and bearing cap data from Bearing #4.

Note the large difference between the shaft and bearing cap direct amplitudes. The shaft vibration waveform plot (Figure 2) shows a definite, repeatable pattern. The shaft vibration half spectrum plot (Figure 3) illustrates two predominant frequencies—a subsynchronous component near 0.5X and the synchronous component at 1X. A component at 1.5X is also visible. The 0.5X, 1X, and 1.5X components in the bearing cap vibration half spectrum plot (Figure 4) occur at amplitudes close to the noise floor, indicating a low signal-to-noise ratio. It is clear from these plots that very little of the shaft vibration was being transmitted to the bearing caps. Data from the Seismoprobe transducers essentially contained no useful information.

Figure 2 shows that the Keyphasor marks on the waveform occur at the same location on each vibration cycle. Thus, the vibration was occurring at an integral submultiple of running speed; if the Keyphasor marks had drifted across the waveform pattern, then the vibration would not have been an integral submultiple. It was clear that the vibration was occurring at *exactly* 0.5X, rather than *about* 0.5X.

Vibration about 0.5X would be a symptom of a fluid whirl or whip instability. Vibration at exactly 0.5X tends to be a symptom of a rub or a loose, oversize, or improperly lubricated bearing (although fluid whirl/whip still cannot be ruled out). Orbit and full spectrum plots would have been useful in this case by providing information on the direction of vibration relative to shaft rotation (forward/reverse precession). A rub can exhibit either forward or reverse precession (depending on its severity), while loose, oversize, or improperly lubricated bearings and fluid whirl/whip exhibit forward precession.

Initial transient analysis

The next step in the analysis was the acquisition of transient data. The compressor train was shut down soon after acquiring the steady state data. Figures 5-7 present proximity data in Bode, polar, and half spectrum cascade plot formats. Transient data from the Seismoprobe transducers provide no useful information and are not presented.

The Bode and polar plots were left uncompensated, since only a small amount of mechanical runout was present at Bearing #4. The presence of significant mechanical runout would have posed a problem when comparing different runs; it is not possible to compensate with a single, fixed runout vector when using the TK16.

The transient data provided the following information:

1) The first balance resonance speed (critical speed) appeared to be at 5850 rpm (Figures 5 and 6). The design, or nameplate, balance resonance speed was reported to be at 6700 rpm. This reduction in the balance resonance speed was attributed to a decrease in the direct stiffness of the rotor/support system.

2) The 0.5X vibration dies out at speeds below 9150 rpm (Figures 5 and 7).

A rub could now be ruled out based on this information. A 0.5X rub occurs at machine speeds *above* twice the first balance resonance speed. In this case, a 0.5X rub might occur at speeds *above* 11,700 rpm; however, the 0.5X vibration was observed as low as 9150 rpm. In addition, rubs tend to *increase*, rather than *decrease*, the system's direct stiffness.

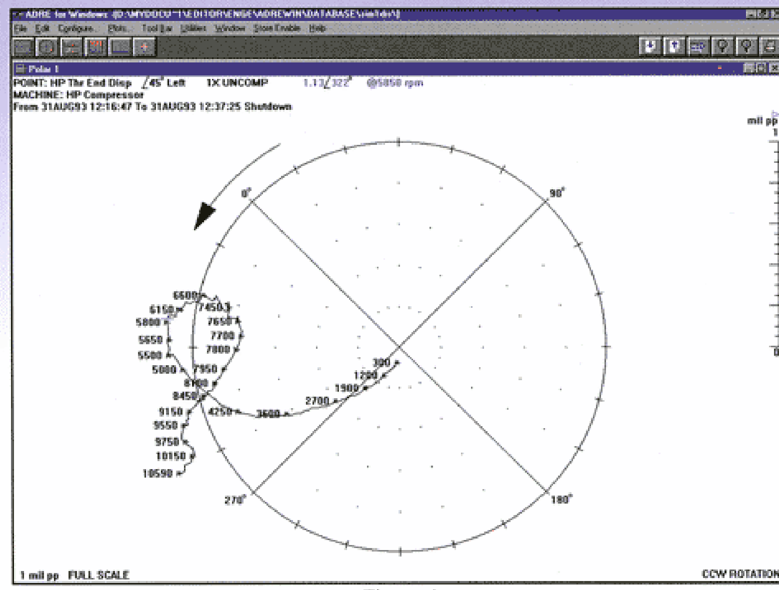

Figure 6
Initial shutdown polar plot, HP compressor thrust end (Bearing #4).

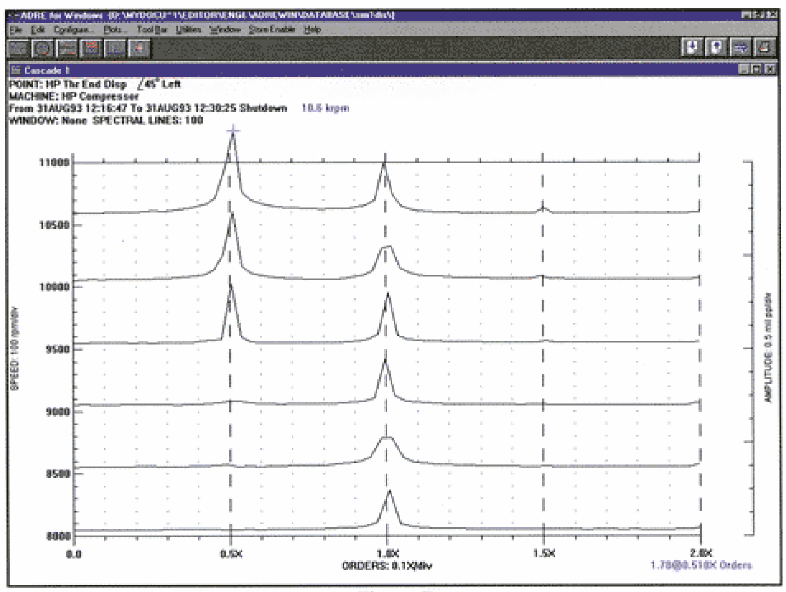

Figure 7
Initial shutdown half spectrum cascade plot, HP compressor thrust end (Bearing #4).

Incorrect bearing insert found

A meeting was called after the initial shutdown to discuss the vibration data and to formulate an action plan. Due to the information provided by the transient data, MMS, plant personnel, and the compressor manufacturer's field representatives decided to inspect Bearing #4 before any other action was taken.

The plain-bore bearing insert was found to be incorrectly manufactured. Correctly manufactured inserts have an oil groove in the center of the insert with an 0.003 inch greater diametral clearance. The bearing insert that was initially in place was found to have a center clearance identical to the edge

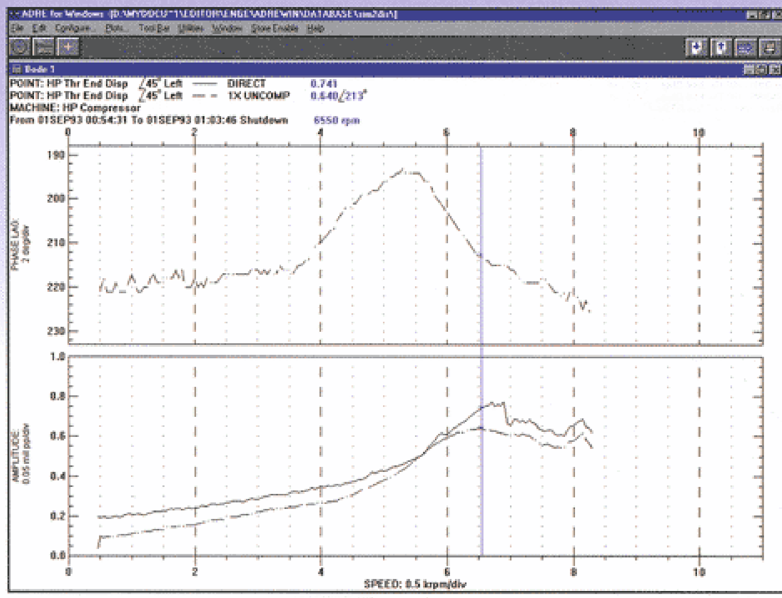

Figure 8

Shutdown Bode plot (8300 to100 rpm), HP compressor thrust end (Bearing #4).

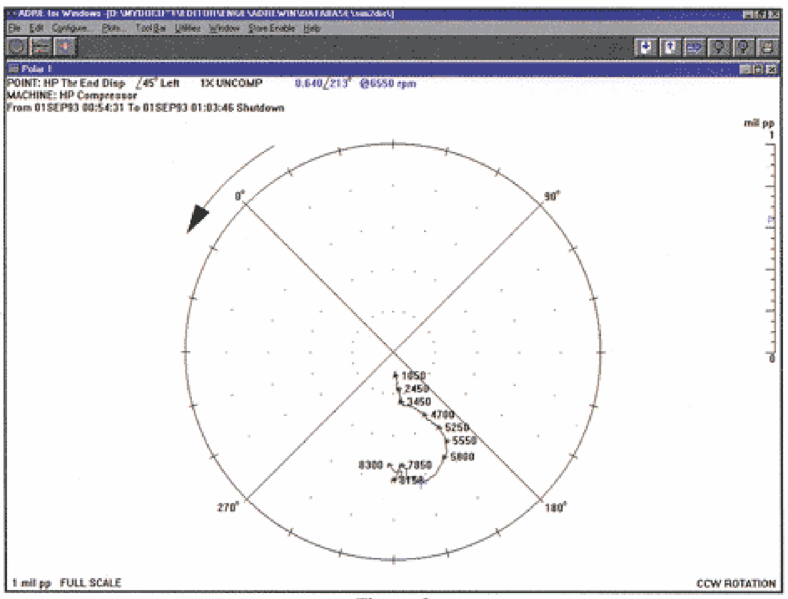

Figure 9

Shutdown polar plot (8300 to 100 rpm), HP compressor thrust end (Bearing #4).

clearance; there was no oil groove. A correct bearing insert was located and installed.

After the bearing insert was replaced, we decided to restart the compressor train rather than continue disassembly. This decision was based on the fact that we had discovered a mechanical problem whose symptoms were evident in the acquired information. This action was taken only after careful consideration, since the plant process cycle had to be restarted in order to run the compressor train above 9000 rpm, where the 0.5X vibration occurred. It was important to identify the root cause of the problem immediately, since each restart over 9000 rpm entailed a certain amount of time and preparation.

Final transient analysis

The compressor train was restarted and brought up to 8300 rpm, shut down, and restarted and brought up to 10,599 rpm. Figures 8 and 9 document the 8300 to 100 rpm shutdown; Figures 10 and 11 document the second restart from 7000 to 10,599 rpm. Please note that these plots are from two separate runs. The numerical values of the phase readings are not consistent between runs because the TK16 Multiplier/Divider selected a different phase reference point for each run. Phase data from Figures 8 and 9 should not be directly compared to phase data from Figures 10 and 11.

Figures 10 and 11 illustrate the decrease in the direct and 1X amplitudes which occurred above 9000 rpm after replacement of the bearing insert. There was no significant 0.5X vibration during this last restart. The final steady state shaft direct amplitudes at 10,595 rpm were 5.6 μm (0.22 mil) pp at the non-thrust end (Bearing #3) and 30.0 μm (1.18 mil) pp at the thrust end (Bearing #4). Figures 8 and 9 indicate that the first balance resonance speed shifted upwards, with the 1X uncompensated amplitude now peaking at 6550 rpm. The new balance resonance speed closely matched the nameplate balance resonance speed of 6700 rpm.

The observed phenomena could now be explained. It was likely the original, non-relieved bearing insert was improperly lubricated, which resulted in a fluid-induced instability. After it was replaced, the as-designed lubrication properties

were restored. The second bearing had less surface area compared to the first bearing (due to the oil groove). This increased bearing loading and, thus, the direct stiffness. The increased load would also tend to stabilize the bearing. Note that the original bearing insert had the same diametral clearance as the replacement bearing insert; it was not oversized.

NOTE: *The phase lag data in plots 8 & 9 and 10 & 11 are from different runs and cannot be compared because of the way the phase reference (Keyphasor) signal was generated.*

The 1X phase lag angles are measured using a Keyphasor signal and the filtered 1X waveform. The Keyphasor signal for these databases, though, was created from a 60X signal generated by a 60 tooth gear. A Bently Nevada TK16 instrument converted the 60 pulse per revolution to 1 pulse per revolution with a divider circuit. Since no single reference tooth can be used to trigger the divider circuit, the 1X phase lag angle information for each run of the machine cannot be compared. It also limits the ability to present compensated 1X plots.

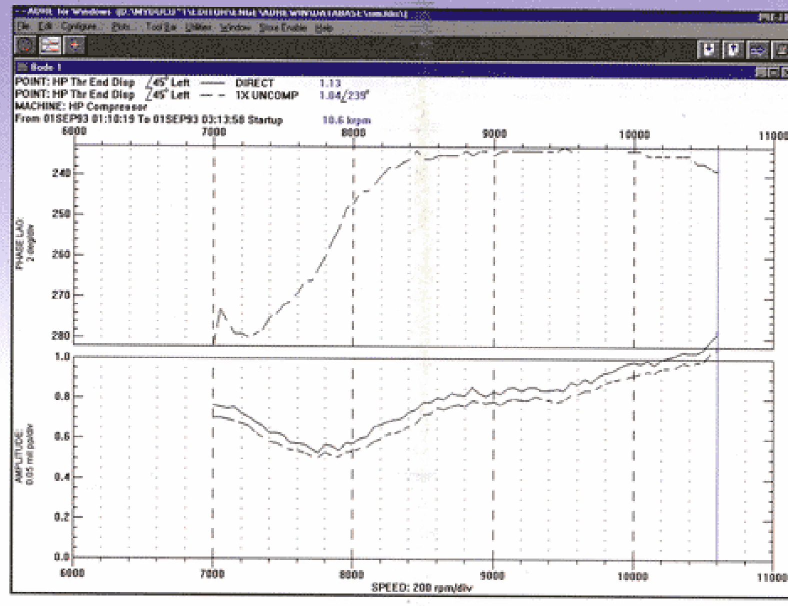

Figure 10
Restart Bode plot (7000 to 10,599 rpm), HP compressor thrust end (Bearing #4).

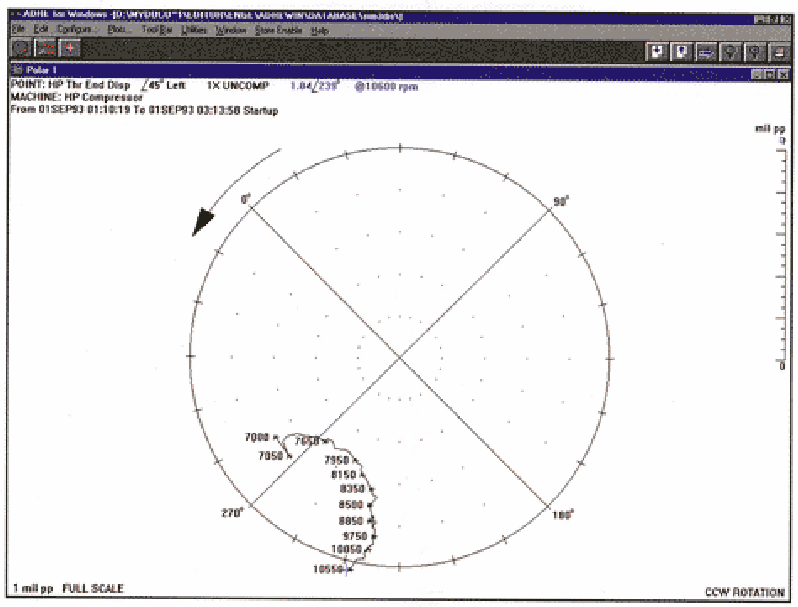

Figure 11
Restart polar plot (7000 to 10,599 rpm), HP compressor thrust end (Bearing #4).

Conclusion

A full toolbox and the acquisition of transient data can enable the diagnostician to quickly and accurately determine a machine's problem and its root cause. "Good luck" cannot be trusted to replace a full toolbox. The root cause of the 0.5X vibration was not positively identified until the final series of startups and shutdowns. The capability for orbit, shaft centerline, and full spectrum analysis would have been very helpful in diagnosing the improperly-lubricated bearing installed on the high pressure compressor.

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